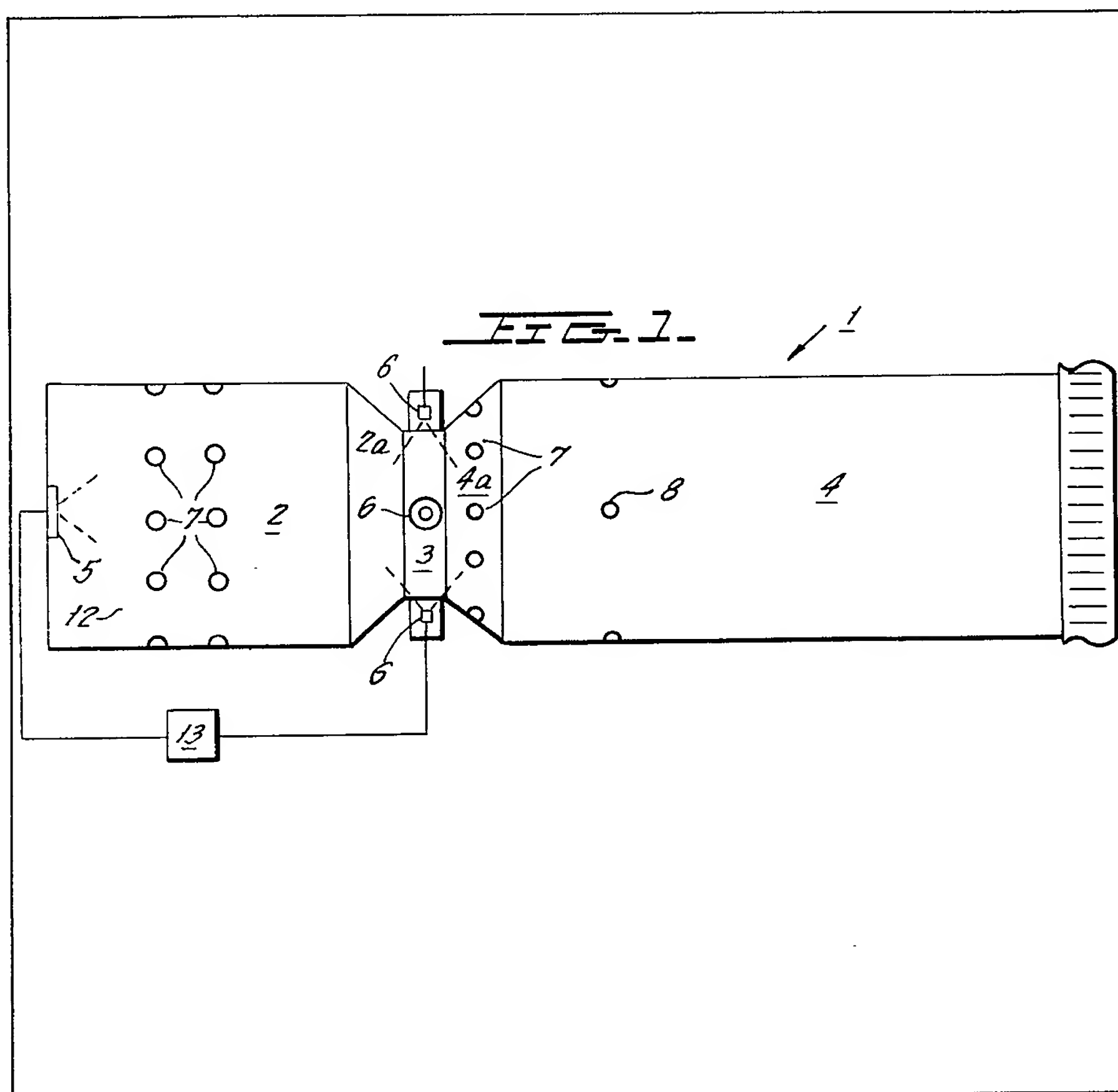


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(54) **Dual stage-dual mode low emission gas turbine combustion system**

(57) A gas turbine combustor arrangement comprises two combustion chambers 2, 4, separated by a necked down throat portion 3 with provision 5, 6, for fuel introduction into each of the chambers. Initially, fuel is introduced into the first chamber and ignited. Thereafter, fuel is introduced into the second chamber until the total fuel flow to the combustor is at the desired rate. Burning in the first chamber is then extinguished by shifting fuel flow to the second chamber and after burning termination, the fuel distribution is reversed until a low emission operation is realized.



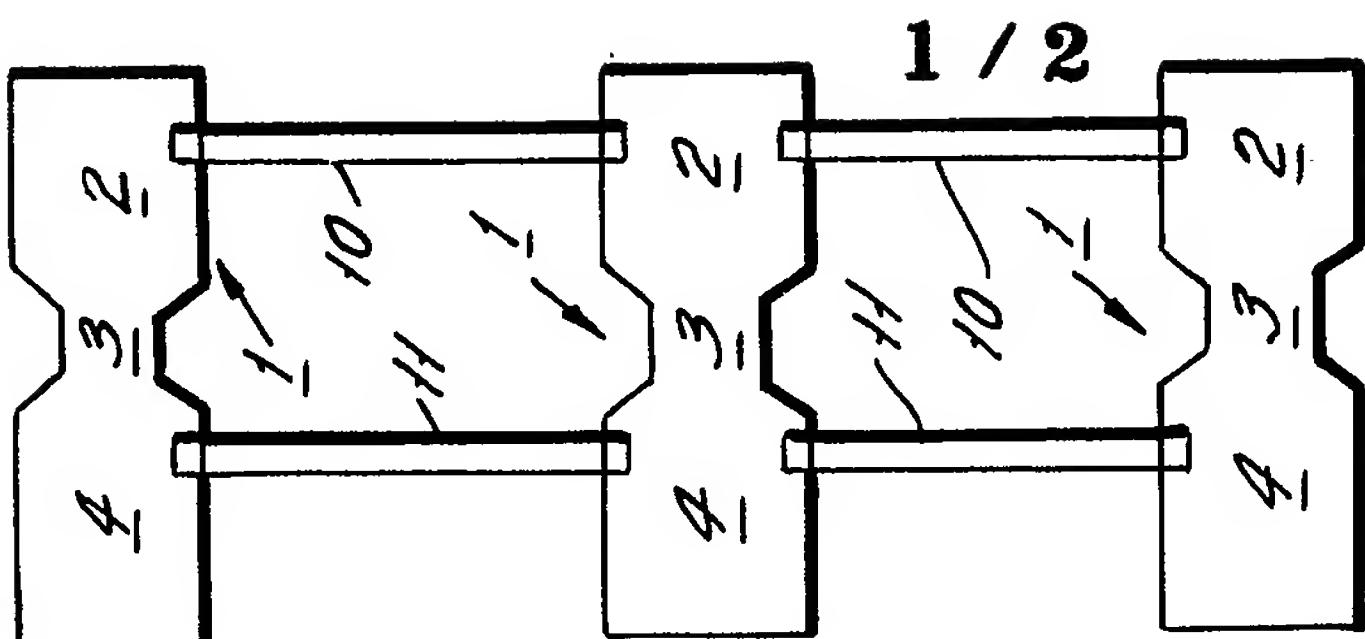


FIG. 3.



FIG. 1.

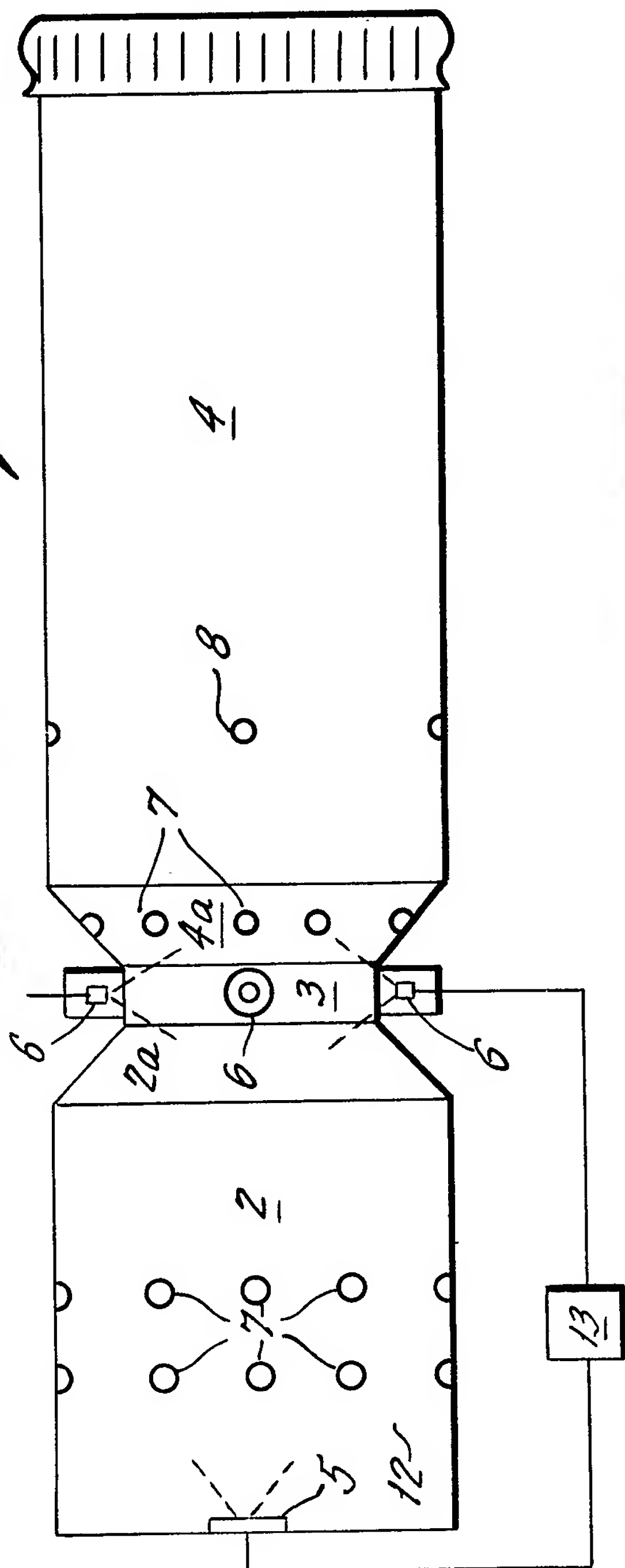


FIG. 2.

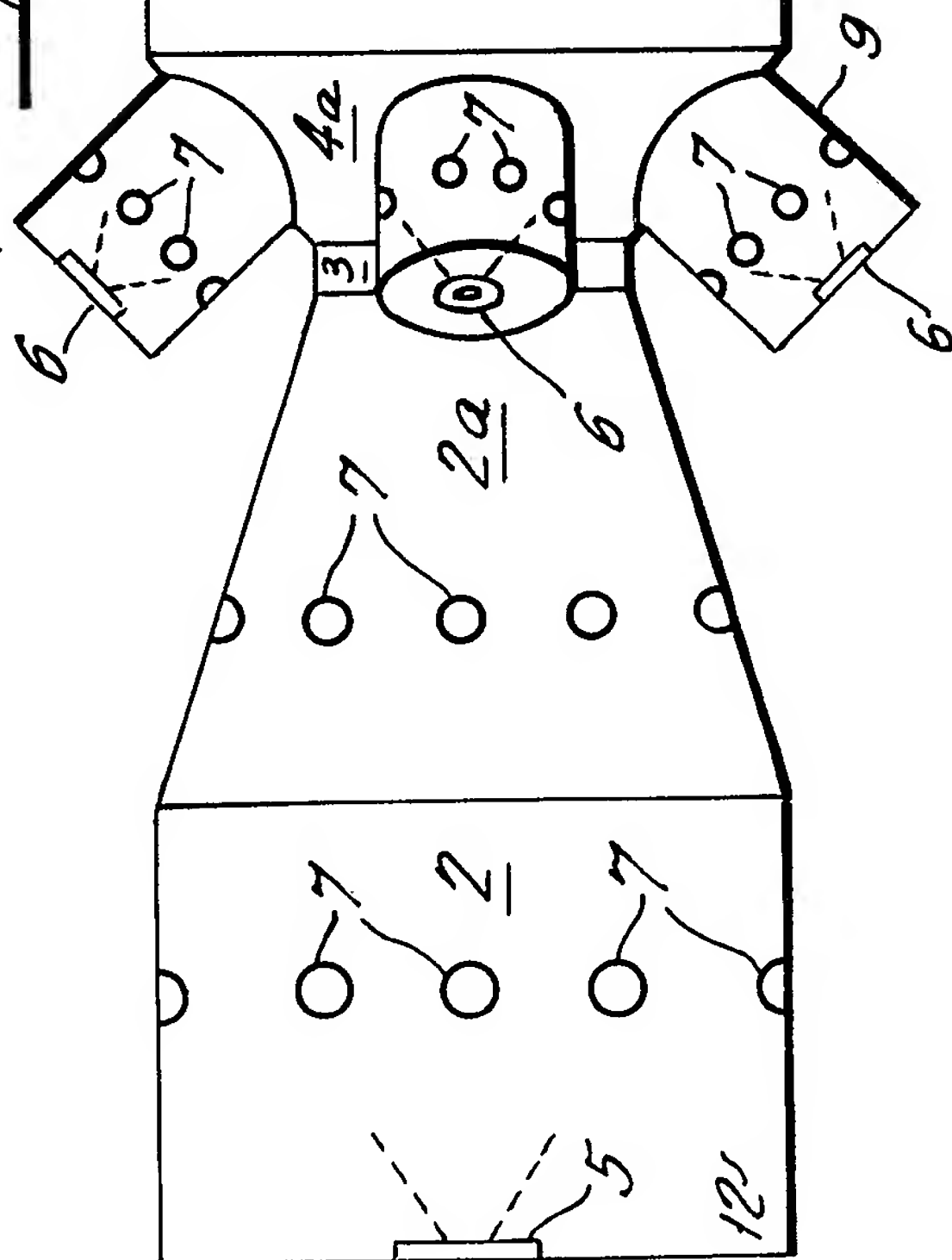
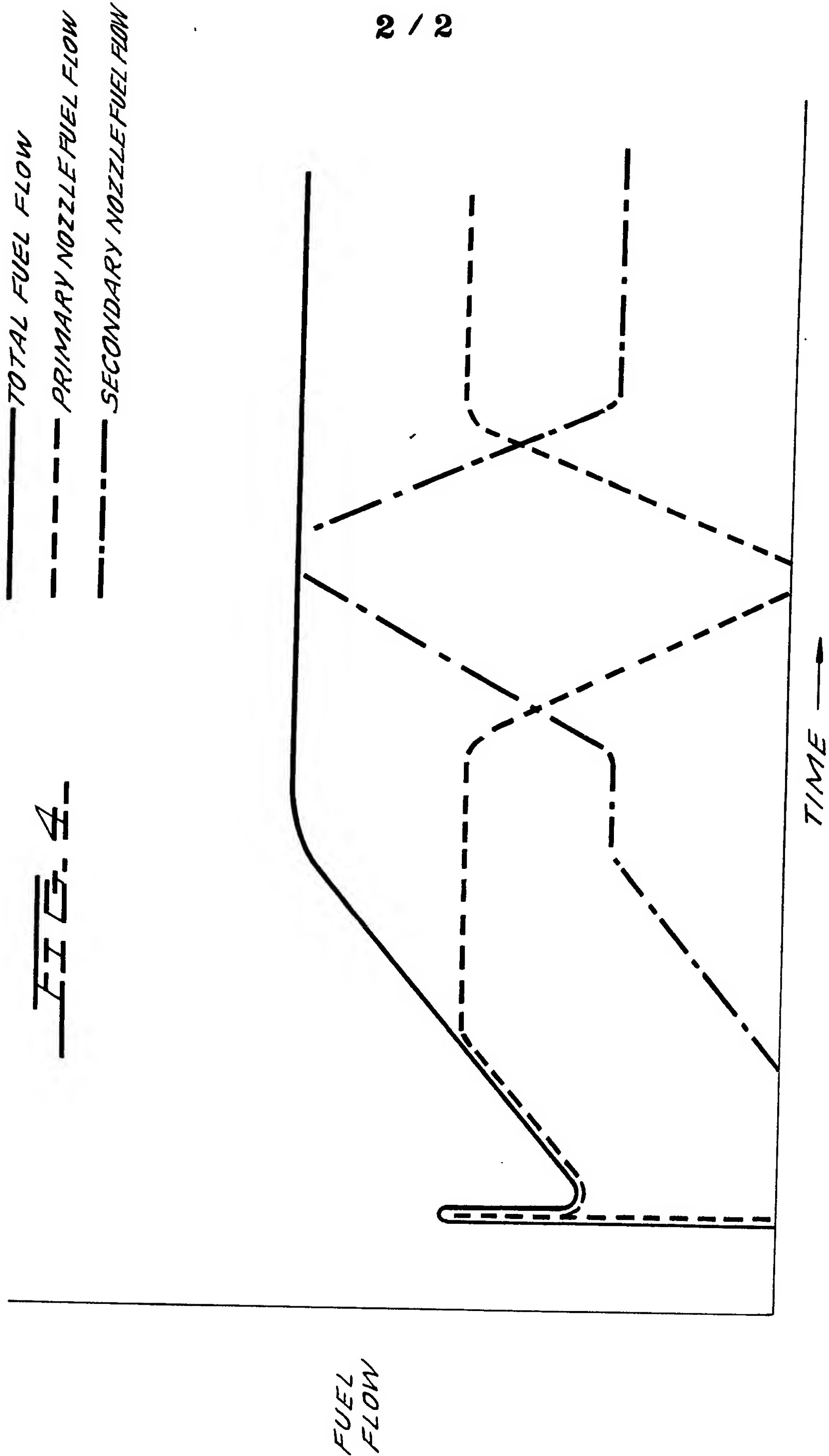


FIG. 4



SPECIFICATION

Dual stage-dual mode low emission gas turbine combustion system

In recent years, gas turbine manufacturers have become increasingly concerned with pollutant emissions. Of particular concern has been the emissions of nitrogen oxides (NO_x) because such oxides are a precursor to air pollution.

It is known that NO_x formation increases with increasing flame temperature and with increasing residence time. It is therefore theoretically possible to reduce NO_x emissions by reducing the flame temperature and/or the time at which the reacting gases remain at the peak temperatures. In practice, however, this is difficult to achieve because of the turbulent diffusion flame characteristics of present day gas turbine combustors. In such combustors, the combustion takes place in a thin layer surrounding the evaporating liquid fuel droplets of a fuel/air equivalency ratio near unity regardless of the overall reaction zone equivalence ratio. Since this is the condition which results in the highest flame temperature, relatively large amounts of NO_x are produced. As a result, the conventional single-stage single-fuel nozzle spray atomized combustors may not meet newly established emission standards no matter how lean the nominal reaction zone equivalence ratio.

It is known that the injection of significant amounts of water or steam can reduce NO_x production so that the conventional combustors can meet the low NO_x emission requirements. However, such injection also has many disadvantages including an increase in system complexity, an increase in operating costs due to the necessity for water treatment, and the degrading of other performance parameters.

Attempts to achieve a homogeneous lean reaction zone by externally prevaporizing and premixing fuel and air at lean equivalence ratios have only limited applicability. These designs have typically been used for clean, very volatile fuels such as gasoline, jet fuel, etc., for regenerative cycle (elevated combustor inlet temperature), and at reduced pressures (less than 10 atmospheres). Beyond the increase in complexity, a serious drawback to this approach is the danger of autoignition and flashback. At 10 atmospheres pressure, the residence time required for complete vaporization of distillate fuel and that for autoignition is nearly the same. See, e.g., ASME Preprint 77-GT-69.

The problem of realizing low NO_x emissions develops further complexity when it is necessary to meet other combustion design criteria. Among such criteria are those of good ignition qualities, good crossfiring capability, stability over the entire load range, large turndown ratio, low traverse number, long life and the ability to operate safely.

Some of the factors which result in the formation of nitrogen oxides from fuel nitrogen and air nitrogen are known and efforts have been made to adapt various combustor operations in light of these factors. See, for example, United States Patents 3,958,416, 3,958,413 and 3,946,553. The processes

used heretofore, however, have either been not adaptable for use in a combustor for a stationary gas turbine or have been inadequate for the reasons set forth below.

It is the object of this invention to provide a new dual stage-dual mode combustion system for a gas turbine which will operate over the entire gas turbine cycle at flame temperatures which will substantially reduce pollutant emissions to acceptable levels using various gaseous and distillate fuel. This and other objects of the invention will become apparent to those skilled in this art from the following detailed description in which:

Figure 1 is a schematic cross-section of a first embodiment of the present invention;

Figure 2 is a schematic cross-section of a second embodiment of the present invention;

Figure 3 is a schematic representation of three combustors of the present invention having a high load ignition system; and

Figure 4 is a graph illustrating the fuel flow in the operation of the present combustors as a function of time.

This invention relates to a combustor for a stationary gas turbine. For example, an arrangement of combustors and the method by which they are operated in order to realize a reduction in NO_x emissions. More particularly, the combustor has two combustion chambers connected through a neck portion, separate fuel introduction means for each section and means to regulate the flow of fuel of each fuel injection means relative to the others. In case blowout should occur, the combustors are provided with a high load ignition system by connecting each first section of adjacent combustors and each second section of adjacent combustors by crossfire tubes. The combustor is operated by first introducing fuel only into the first section and causing it to burn therein. Thereafter the flow of fuel is shifted into the second section until burning in the first section terminates followed by a reshifting of the fuel distribution into the first section for mixing purposes until the desired NO_x reduction has been achieved.

Referring to Figures 1 and 2, the combustor 1 of the present invention generally comprises a first combustion zone or section 2 which is connected to a neck or throat section or zone 3 which, in turn, is connected to a second combustion zone or section 4.

First combustion zone 2 can be of a conventional lean combustor design utilizing a single, preferably axisymmetric fuel nozzle 5. The second combustion zone 4 is supplied with fuel from a plurality of fuel nozzles 6. In Figures 1 and 2, four radial nozzles located symmetrically on the combustor circumference are shown but any number of nozzles can be used as desired. Air from the gas turbine compressor (not shown) is introduced into the combustor at elevated pressure, typically from about 10-30 atmospheres. For example, the air can be introduced through one or more air entry ports 7. Ports 7 located in first combustion zone 2 are preferably positioned so as to cause a flow recirculation which results in a stable burning over a wide operating range. Provisions is made for the rapid cooling of the combustor

tion products in zone 4 with a suitable heat exchange fluid. For example, quenching air can be admitted to zone 4 through a plurality of apertures 8. The amount of heat exchange fluid employed is that
 5 sufficient to cool the combustion products so as to reduce the fluid temperature to the desired gas turbine firing temperature.

Zones 2, 3 and 4 are preferably of circular cross-section but any desired configuration can be employed. The material of construction can be metal or ceramic and the zones can be surface cooled by a variety of techniques including water-cooling, closed system cooling, steam film cooling and conventional air film cooling. By way of example only, a useful
 10 arrangement of annular rows of schematically spaced louvers along the zone walls to provide air film cooling is described in Dibelius and Schiefer U.S. Patent 3,777,484, and a useful arrangement of slot cooling is described in Corrigan and Plemmons
 15 U.S. Patent 3,728,039.

It will be appreciated that neck of throat 3 acts as an aerodynamic separator or isolator between the first combustion zone 2 and the second combustion zone 4. In order to adequately serve this function,
 25 neck 3 must have an adequately reduced diameter relative to first zone 2 and second zone 4. In general, a ratio of the smaller of the first combustion zone 2 or second combustion zone 4 diameter to neck zone 3 diameter of at least 1.2:1, and preferably at least
 30 about 1.5:1, is employed. To facilitate a smooth transition between first combustion zone 2 and neck 3, the downstream most portion 2a of zone 2 is of uniformly decreasing diameter, i.e., has a conical cross-section. The longitudinal length of neck 3 is
 35 not critical and any distance which will accomplish the separation function and throttling function of neck 3 can be employed. In general, the longitudinal length of the first combustion zone 2 is at least about three times that of neck 3, and preferably at least
 40 about five times that of neck 3. Second combustion zone 4 has the same general configuration at first zone 2 except, of course, that the transitional cone-shaped portion is in the upstream most portion 4a or zone 4 meeting neck 3.

A second and preferred embodiment of the present invention is shown in Figure 2 in which the same reference numerals have been used to designate like parts in Figure 1. The arrangement shown in Figure 2 differs from that shown in Figure 1 in the
 50 following respects. First, the diameter of throat 3 has been reduced in order to increase the average air velocity through the zone, which design is more effective in preventing flashback. The height (i.e. longitudinal length) of convergent conical section 2a
 55 has also been increased. In this embodiment, fuel nozzles 6 have been moved from throat 3 to the divergent conical section 4a of second zone 4 and have been set back in mini combustion chambers or swirler cups 9 where the operation of secondary fuel
 60 nozzles 6 is more stable and it is less likely to experience blowout during the fuel switching procedures described below.

Figure 3 shows by way of example three joined combustors of the present invention. The first combustion zone 2 of each combustor 1 is intercon-

nected with the first combustion zone 2 of the adjacent combustors 1 by means of a crossfire tube 10 in the conventional manner. Additionally in the present invention, second combustion zone 4 of each combustor 1 is interconnected to the second combustion zone 4 of each adjacent combustor 1 through a crossfire tube 11. As will be described below, at the design high load conditions of operation of the present combustors, burning is effected
 70 only in second zone 4 and no burning occurs in first zone 2. If for some reason one chamber blows out under such high load conditions, crossfiring cannot occur in conventional arrangements since the standard crossfiring tubes 10 are located upstream of the
 80 reaction zone 4 and neck 3 serves to prevent flashback. In the embodiment shown in Figure 3, the second set of crossfire tubes 11 act as a high load ignition system. Although it is preferred to provide the dual set of crossfire tubes (i.e., tubes 10 and 11), any high load relight system can be incorporated into the combustor system if desired.

The operation of the combustors of the present invention is shown graphically in Figure 4. Combustion begins by igniting a mixture of a hydrocarbon
 90 fuel and air in first combustion zone 2. This is accomplished in a conventional manner by means of a spark plug 12 which is located near fuel nozzle 5 in first combustion zone 2. In typical conventional installations, ten combustors are arranged in a ring and usually only two of the combustors are provided with spark plugs 12 while the remaining eight combustors are ignited by crossfiring through crossfire tubes 10. During ignition and crossfiring, and also during the low load operation of the combustor,
 95 only the primary fuel nozzle 5 delivers fuel to combustor 1. Up to this point, combustion is a single-stage heterogeneous, turbulent diffusion flame burning characteristic of conventional combustors.

At some mid-range load condition, the exact timing of which is related to stability limits and the pollutant emission characteristic of each mode and the fuel split between stages, the secondary fuel nozzles 6 are activated. Passage of the ignited fuel from first zone 2 into second zone 4 causes ignition in second zone 4. The combustor is now operating in a two-stage heterogeneous mode which continues until the desired base load is achieved. After allowing a short period for stabilization and warm-up, the
 110 operation is converted from a two-stage heterogeneous combustion to a single-stage homogeneous combustion. This procedure begins by simultaneously increasing the amount of fuel to the secondary nozzles 6 and decreasing the amount
 115 to the primary nozzle 5 while the total fuel flow remains constant. The relative rates of fuel flow to nozzles 5 and 6 can be controlled by a fuel flow controller 13 which is interconnected to nozzle 5 and nozzles 6. The change in fuel distribution continues
 120 until the flame goes out in the first combustion zone 2 which, in most instances, is when all of the fuel flow has been transferred to secondary nozzles 6.

Fuel flow to nozzle 5 is then reinitiated or increased and flow to nozzles 6 decreased while
 130 maintaining the total fuel flow substantially con-

stant. Combustor 1 is designed not to flashback under normal operation by making first zone 2 long enough so that the flow cross-section is similar to that of a fully developed turbulent pipe flow and the throat 3 narrow enough so that the velocity is increased to a level above which the flame speed cannot be overcome. As a result, the majority of the fuel and air premix in the first stage (i.e. first zone 2) and combust homogeneously in the second stage, i.e. second zone 4. The switch of fuel distribution from secondary nozzles 6 to primary nozzle 5 continues until the desired low pollutant emission levels are met. The desired levels are achieved when the majority of fuel flow is through nozzle 5 and in most instances, at least 60% of such flow is through nozzle 5.

It should be appreciated that an important feature of the combustor of the present invention is that if flashback should occur, it is not a hardware catastrophe as in typical premixed designs. However, a significant NO_x penalty would result and control steps must be taken to go through the switching procedure again and resume operation in the homogeneous mode.

During shutdown of the gas turbine, steps are taken to relight first zone 2 because there is only a small turndown ratio in the homogeneous mode. Relighting the first stage means that there is a return to the heterogeneous two-stage combustion where the system has a wide turndown ratio allowing the turbine to be brought down slowly to alleviate undesirable thermal stresses.

In order to demonstrate the reduction in NO_x emissions achieved by the present invention, a combustor constructed in accordance with the present invention was compared to a conventional commercially available combustor using MS 7001E equipment. The combustor of this invention had the configuration shown in Figure 1 and utilized a single air atomized MS 7001E nozzle as the primary nozzle 5 and four smaller pressure atomized secondary nozzles 6. Data was collected at about 2080°F, laboratory equivalent to base load, (corrected for radiation losses from thermocouples). Under these conditions, the standard conventional combustor exhibited an NO_x emission in the laboratory of 120 ppmv while a combustor constructed in accordance with the present invention emitted only 56 ppmv. This test was run using a vitiated air supply, which means that the products of combustion from a direct heater (such as a propane heater), used to increase air temperature to proper inlet levels, is utilized as the oxidant for combustion during the tests. Therefore, the NO_x emissions are lower than would be obtained with non-vitiated air. Based on these laboratory results it is expected that operation of the combustor of the present invention under field conditions (i.e., actual turbine use with non-vitiated air) with homogenous operation would exhibit a comparable reduction NO_x emissions. Therefore, it is estimated that combustors constructed in accordance with the present invention will meet low NO_x emission requirements.

A second test of the dual stage/dual mode combustion system of the present invention was con-

ducted during which a vitiated air supply was utilized and during which the firing temperature remained constant at approximately 2070°F. At a point during the increase of fuel flow in the secondary fuel nozzles 6 when the amount of fuel flow through primary nozzle 5 was 20% and there was combustion in both combustion zone 2 and combustion zone 4, the NO_x emission was about 95 ppmv. After switching from the two-stage heterogeneous combustion mode to the homogeneous combustion mode, at a point where approximately 14 percent of the fuel was flowing through the primary nozzle (of the first stage), the NO_x emissions were 93.5 ppmv. The amount of fuel flowing to the primary nozzle 5 was then increased from 14% to a point at which approximately 70% of the total fuel flow was through the primary nozzle and the NO_x emission continued to decrease from 93.5 ppmv to about 49 ppmv.

A third test was carried out in a manner similar to the first test described above but using a non-vitiated air supply, i.e., indirectly preheated air with no combustion products. At a firing temperature of about 2060°F, the conventional combustor emitted about 260 ppmv of NO_x while the combustor of the present invention operating in a homogeneous mode emitted about 65 ppmv. The fuel used in each of the above tests was No. 2 distillate.

From the foregoing laboratory test data and in particular that of the third test utilizing a non-vitiated air supply, those skilled in the art can appreciate the significant reduction (a factor of four) in NO_x emissions achieved by the combustor constructed in accordance with the present invention. By utilizing such combustors, NO_x emission levels will be substantially reduced and will meet most NO_x emission requirements.

Having thus described two embodiments of the present invention and their modes of operation, those skilled in the art can better understand how the invention is distinguishable from the aforementioned prior art patents. Specifically, U.S. Patent 3,946,533 to Roberts et al appear to describe a combustor with two stages and multiple fuel nozzles for emission control. However, the fuel and air are mixed outside the combustion liner wall which is distinguishable from the invention described here. Also, in accordance with the combustor of the present invention, there are some conditions where the reaction occurs in an unpremixed heterogeneous mode (i.e., during startup, part load and transient periods of base load), a mode of operation not possible in the combustor of the Roberts et al patent. The modes of operation of the present invention facilitate a large turndown ratio, easy ignition and crossfiring, and flame stability, essential characteristics of a practical design. Also, switching from the heterogeneous to the homogeneous mode of operation is achieved in accordance with the present invention by varying the fuel split between the first and second stage fuel nozzles, a characteristic not disclosed by Roberts et al.

U.S. Patents 3,958,413 to Cornelius et al and 3,958,416 to Hammond, Jr. et al relate to two-stage combustors with the stages separated by a converging-diverging throat section. Also, the first stage of

both of these patents is used at some times during the cycle as a section where combustion occurs and at other times in the cycle where premixing occurs. Therefore, flashback does not cause a hardware catastrophe, as would be the situation in the Roberts et al patent. The Cornelius et al and Hammond, Jr. et al patents also appear to describe a variable air inlet geometry for changing the air scheduling between stages to accomplish the transition from heterogeneous combustion in the first stage or in the first and second stages to homogeneous combustion in the second stage only. In contradistinction, the present invention utilizes fuel scheduling between stages, utilizing multiple fuel nozzles (rather than variable geometry) and varying the fuel split rather than the air split.

CLAIMS

1. A method of operating gas turbine combustor to produce low NO_x emission, said combustor having first and second combustion chambers interconnected by a throat chamber, said first chamber having first fuel introduction means and at least one of said throat chamber and said second chamber having second fuel introduction means, comprising in sequence,
 - (a) continuously introducing fuel into said first chamber through said first fuel introduction means and causing said fuel to burn therein;
 - (b) commencing the introduction of fuel to said second chamber through said second fuel introduction means and increasing the rate of introduction means is about the desired fuel introduction rate, ignition of the fuel in said second chamber being commenced as a result of passage of combustion products from said first chamber to said second chamber;
 - (c) decreasing the rate of fuel introduction through said first fuel introduction means and correspondingly increasing the rate of introduction through said second fuel introduction means such that the overall rate of introduction remains substantially constant until at least the burning of fuel in said first chamber ceases; and
 - (d) increasing the rate of fuel introduction through said first fuel introduction means and correspondingly decreasing the rate of introduction through said second fuel introduction means such that the overall rate of introduction remains substantially constant until the desired level of NO_x emission from the combustor is realized.
2. The method of claim 1 wherein the fuel introduction through said first fuel introduction means in step (c) is decreased until all fuel introduction is through said second fuel introduction means.
3. The method of claim 2 wherein the fuel distribution at the end of step (d) is such that the majority of fuel is introduced through said first fuel introduction means.
4. The method of claim 3 wherein said fuel distribution at the end of step (d) is such that at least 60% of the total fuel is introduced through said first fuel introduction means.
5. The method of claim 1 wherein the fuel

distribution at the end of step (d) is such that the majority of fuel is introduced through said first fuel introduction means.

6. The method of claim 1 wherein said first fuel introduction means is an axisymmetric fuel nozzle and said second fuel introduction means comprises a plurality of fuel nozzles symmetrically disposed.

7. A combustor for a stationary gas turbine capable of producing reduced NO_x emissions upon the burning of a hydrocarbon fuel which comprises a first combustion zone, first hydrocarbon fuel introduction means disposed so as to introduce fuel into said first zone, means to introduce compressed air into said first zone, a neck zone connected to said first zone and having a cross-sectional area less than that of said first zone, a second combustion zone connected to said neck zone and having a cross-sectional area greater than that of said neck zone, means to introduce compressed air into said second zone, second hydrocarbon fuel introduction means disposed so as to introduce fuel into at least one of said neck and said second zone, means to introduce a quenching amount of a heat exchange fluid into said second zone downstream of said means to introduce compressed air into said second zone, and means to alter the relative rates of hydrocarbon fuel flowing through said first and second hydrocarbon fuel introduction means.

8. The combustor of claim 7 wherein said first hydrocarbon fuel introduction means is a single axisymmetric fuel nozzle and wherein said second hydrocarbon fuel introduction means comprises a plurality of symmetrically disposed fuel nozzles.

9. The combustor of claim 8 wherein each of said plurality of fuel nozzles are adapted to introduce fuel into a third combustion zone which contains means to introduce compressed air therein and said third combustion zone, is disposed so as to transfer the fuel and air therein into said second combustion zone.

10. The combustor of claim 9 wherein said first, second and third combustion zones and said neck zone are each of circular cross-section.

11. A stationary gas turbine combustor arrangement comprising a plurality of adjacent combustors, each of said combustors having a first and second combustion chamber interconnected by a neck, at least one combustor having means to ignite a fuel in the first chamber thereof, a plurality of first crossfire tubes each of which interconnects the first chamber of one combustor with the first chamber of an adjacent combustor, and a plurality of second crossfire tubes each of which interconnects the second chamber of one combustor with the second chamber of an adjacent combustor.

12. The stationary gas turbine combustor arrangement of claim 11 comprising ten combustors each of which is adjacent to other combustors.

13. A method as claimed in claim 1 and substantially as described herein with reference to the accompanying drawings.

14. A combustor for a gas turbine substantially as described herein with reference to Figure 1, or Figure 2, of the accompanying drawings.

15. A gas turbine combustor arrangement sub-

stantially as described herein with reference to the accompanying drawings.

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ABSTRACT:

CHG DATE=19990617 STATUS=O> A gas turbine combustor arrangement comprises two combustion chambers 2, 4, separated by a necked down throat portion 3 with provision 5, 6, for fuel introduction into each of the chambers. Initially, fuel is introduced into the first chamber and ignited. Thereafter, fuel is introduced into the second chamber until the total fuel flow to the combustor is at the desired rate. Burning in the first chamber is then extinguished by shifting fuel flow to the second chamber and after burning termination, the fuel distribution is reversed until a low emission

operation is realized. 